EFFECT OF THE DESIGN ARRANGEMENT OF MECHANICAL WRENCHES ON THE PROCESS OF SCREWING TOGETHER AND UNSCREWING PUMP AND COMPRESSOR PIPES

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It was noted in [1] that the torque of a mechanical wrench is expanded on running in the threaded joint (shearing off of microunevennesses, their plastic and elastic deformations) and on inducing the state of stress in the joint determining both the strength properties of the joint and its tightness. However, at a certain application of the forces rotating the nipple relative to the muff, a bending moment is induced in the threaded joint, and this moment has to be overcome in the process of screwing together [2, 3]. Thus the torque developed by the wrench is

\[ M_t = M_r + M_c + M_b, \]

where \( M_r \) is the torque of running in; \( M_c \) is the torque inducing the state of stress in the threaded joint; \( M_b \) is the torque for overcoming the bending moment.

In connection with that a topical task is the maximal reduction of the torques \( M_r \) and \( M_b \). Methods of reducing \( M_r \) were dealt with in [1]. The torque \( M_b \) depends largely on the design of the wrench.

At present four types of wrench are known:

stationary type wrenches with planetary gear (Fig. 1b) where the torque is transmitted by a sun wheel through the machine wrench slipped manually over the pipe (the arrangement of applying the forces is analogous to the arrangement of the forces when two machine wrenches operate);

suspension type wrenches with planetary gear (Fig. 1a) in which the arrangement of application of the forces is analogous to the application of forces in operation with one machine wrench;

suspension type wrenches (Fig. 1c, d) having gripping elements built into the rotor pinion (split, closed, lockable); the wrenches may be provided with either of two kinds of counterrotating attachments: machine-type wrench or a simplified pipe-gripping device (the rotator and the counterrotating attachments are interconnected);

standard-type wrenches (Fig. 1e) with gripping elements built into the rotor pinion; the wrenches are mounted on the column flange and are provided with counterrotating attachments (simplified pipe-gripping devices).
Fig. 1. Design arrangements of wrenches and stress-strain diagrams of the transverse bending moments from the forces $F$ inducing the torque [4]: a, b) suspension-type and stationary-type planetary gear arrangement, respectively; c, d) suspension type with gripping elements with counterrotating attachments type machine wrench and simplified pipe-gripping device, respectively; e) stationary type with counterrotating attachment type simplified pipe-gripping device and gripping elements.

It can be seen from an analysis of Fig. 1 that $M_b$ can be reduced by changing the arms through which the forces inducing this moment are applied. The effect of transverse bending on threaded joints was dealt with in [5-7]. Mamedov [5] and Pesnyak [6] determined the state of stress in a threaded joint by specifying the internal force of interaction between the elements. Mamedov [5] assumed that the pressure on the individual threads is uniformly distributed along the joint; in [7] the stress-strain diagram of the pressure was adopted as triangular. In [6] it was noted that in reality the state of stress of the pipe and muff has to be determined by taking the joint deformation of the elements of the joint into account. In that case, however, we deal with a cylindrical threaded joint under conditions of linear deformation, and the solution is extended to petroleum pipes. The results obtained in [6] can therefore yield only a preliminary notion of the mechanism of interaction between the elements of the joint in bending. The publications [5, 7] also contained photographs of the section of a threaded joint subjected to bending which clearly shows the load distribution along the thread, i.e., between the individual turns (in the form of a triangle). On account of the bending moment the nipple has linear contact with the muff during rotation [5, 7]; therefore, in an arbitrary cross section of the joint the running pressure can be represented in the form [5]

$$p = p_1 \cos \frac{\varphi}{2},$$

where $p_1$ is the normal pressure; $\varphi$ is the angle between $p_1$ and $p$.

In the described section of the threaded joint of pump and compressor pipes there arise axial tensile, radial and tangential compressive stresses under the effect of the bending moment. The tensile stress can be determined by the known formula

$$\sigma_t = \frac{M}{W},$$

where $M$ is the maximal bending moment; $W$ is the moment of resistance of the weakest cross section of the thread.

To determine the stresses in the meridional section at the distance $X$ from the point 0 we isolate an element with width $dX$ whose area is equal to

$$\frac{dx}{\cos \alpha}r d\varphi.$$  [5] (here, $\alpha$ is the profile angle of the thread; $r$ is the running value of the radius of the thread).

Multiplying the values of these areas by the values of the normal pressures in the corresponding sections, we find the elementary forces whose sum is

$$\sum_{i=1}^{n} p_i \frac{dX}{\cos \alpha} r d\varphi \cos \frac{\varphi}{2}.$$